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<https://doi.org/10.23939/tt2022.01.065>

INFLUENCE OF VEHICLE ACCELERATION INTENSITY ON DUAL-MASS FLYWHEEL ELEMENTS AND TRANSMISSION LOAD

Summary. *Modern high-torque low-speed internal combustion engines (ICEs) generate torsional vibrations close in disturbance frequency to gearboxes natural oscillation frequencies. Effective absorption of such oscillations requires a new torsional vibration damper between the internal combustion engine and gearbox design, which is implemented in the form of a dual-mass flywheel (DMF). One of the main reasons for DMF failure is its spring components destruction. The article develops mathematical and simulation (in MATLAB Simulink environment) model of a car with DMF in the period of starting, which takes into account the dependence of torque and power of the internal combustion engine on the number of the crankshaft revolutions and uneven rotation, car inertial and stiffness parameters, road resistance. It is established that when the car starts in first gear, the maximum load on spring components of DMF and transmission occurs at the initial moment of clutch engagement and exceeds the maximum effective torque of the internal combustion engine 1.6 times, has a pronounced oscillatory character and stabilizes as the car accelerates. With smooth acceleration of a car, when torque of internal combustion engine reaches, but does not exceed its maximum value of 250 N·m, elastic moment in transmission components is stabilized at 230 N·m. During intensive acceleration and transition through the extremum on torque curve of internal combustion engine on number of crankshaft revolution, the maximum DMF spring components and transmission load initially doesn't change significantly, but reduces the duration of oscillatory processes and elastic moment of 160 N·m after attenuation of oscillations. A similar nature of stress changes is observed in the elastic links of DMF, which eventually leads to their fatigue failure and DMF failure.*

To increase a DMF service life, it is advisable to accelerate a car when moving intensively, bringing a number of revolutions to a value that is located at the extreme of torque of internal combustion engine on its performance characteristic, followed by switching to the next gear.

Key words: *transmission, dual-mass flywheel, dynamic model, mathematical model, simulation model, torque.*

1. INTRODUCTION

Strict environmental standards, which are now imposed on cars with internal combustion engines (ICE), have led to development and use in cars of high-torque low-speed internal combustion engines that generate torsional vibrations close in disturbance frequency to gearboxes natural frequencies. Effective absorption of such oscillations requires a new design of the torsional vibration damper between an internal combustion engine and a gearbox, which is implemented in the form of a dual-mass flywheel (DMF) [1].

2. RESEARCH RELEVANCE

The cost of DMF in 2 – 6 times, depending on the design, exceeds the cost of a conventional single-mass flywheel. DMF don't cause special claims only to 80 – 100 thousand km, after which the risk of failure increases significantly. One of the main reasons for a DMF failure is the destruction of its spring

components [9, 11]. Since the style of driving, and especially abrupt movement in first gear or maneuvers during overtaking, significantly affect the load of both transmission and DMF spring components, the study of the impact of transient modes of movement on the load of DMF sprung components, in order to clarifying and eliminating the causes of their premature failure is relevant and of great practical importance.

3. PROBLEM FORMULATION

The main task of a DMF is absorption of torsional vibrations of the internal combustion engine crankshaft at low speeds, in particular at idling, when the frequency of forced oscillations approaches the gearbox oscillations natural frequency primary shaft and gears located on it. The stiffness of the DMF spring components should be lowered to maximize torsional vibrations energy generated by the internal combustion engine. With increasing crankshaft speed and, accordingly, the torque of the load on the DMF sprung components increases and for its perception the stiffness of DMF sprung components must increase. Accordingly, there are different DMF designs: with twisted springs of constant diameter, springs of different stiffness, separated by separators, etc. [1]. Despite a number of advantages, the disadvantage of the latter is that soft springs break down the fastest, which leads to the loss of DMF and requires its replacement.

As DMF repair is not provided by manufacturers, but requires its replacement together with the clutch and its components, the problem of increasing its service life needs to be solved by design changes or develop recommendations for the operation of vehicles, including unsteady.

4. AIMS AND OBJECTIVES OF THE STUDY

The aim of the work is to develop simulation models and study the impact of oscillating processes in the drive of a car with DMF on the load of its parts during intensive acceleration of the car when starting from the first speed, development of recommendations for increasing DMF lifespan.

5. ANALYSIS OF RECENT RESEARCH AND PUBLICATIONS

A number of works are devoted to the study of oscillatory phenomena in car transmission a with a DMF and optimization of their design parameters [2–14].

Thus, in [2] a generalized model of a DMF with bifilar centrifugal pendulum vibrational absorber (CPVA) is proposed. Taking into account this model, a dynamic model of the drive of a car equipped with DMF with bifilar CPVA is built. The dynamic reactions of the system in the conditions of idling and car movement are investigated. It is shown that, compared to DMF with conventional centrifugal pendulum vibration absorber, DMF with bifilar type CPVA has better vibration damping in the entire range of engine crankshaft speeds.

In [3] a mathematical model of a DMF based on the equation of motion of internal combustion engine and DMF masses was developed. The peculiarity of the work is to take into account the processes occurring in the cylinders of the internal combustion engine on different operating cycles, and their consideration in describing the change in effective torque generated by the internal combustion engine. To simplify, the moment of resistance created by the transmission was considered constant. The influence of DMF on the efficiency of its absorption of oscillations generated by internal combustion engines and their transmission to the transmission components has been studied.

In [4] the results of theoretical modeling and experimental study of elastic-dissipative characteristics of a DMF taking into account the frictional interaction of the twisted spring links with the flywheel body are presented. It is established that the torsional stiffness of a DMF varies depending on the amplitude and frequency of perturbation.

In [5] it is proposed to equip a DMF to increase its efficiency with an electromagnetic torsional vibration damper with the ability to adjust its characteristics. According to the author, the new electric dual-mass flywheel (EDMF) has better characteristics than the classic passive DMF.

In [6] on the basis of the developed model of the drive of the car with DMF influence of kinetic parameters of a flywheel on torsional fluctuations of elements of transmission in the course of engine start is investigated. The test and simulation results showed that a DMF undergoes strong oscillations when the transmission passes through the resonant zone. The design requirements for a DMF at engine start are obtained, in particular, it is recommended to observe the ratio between the moments of inertia of primary and secondary masses in the range of 0.7–1.1, and to choose the minimum stiffness of springs that perceive oscillations when starting the engine.

The work [7, 8] is devoted to the issue of optimization of design parameters of DMF for high-load car powertrains.

Analyzing the standard driving cycle of the car in urban conditions, we can see that the DMF sprung components perceive the maximum load on non-stationary modes of movement of the car (moving, overtaking maneuvers, sharp acceleration). Under such conditions, soft DMF springs, which absorb the oscillations of the crankshaft of the internal combustion engine during its start-up or idling, in non-stationary modes of movement of the car operate at the allowable stresses and break down the fastest [9].

The article [10] investigated the process of moving the car from the ground, but the magnitude and nature of the change in torque of the internal combustion engine was set by a harmonic function that did not take into account the change in torque from the angular velocity of the crankshaft.

6. PRESENTING MAIN MATERIAL

Car powertrain model with a DMF. We present a generalized dynamic model of the car powertrain with a DMF in the form shown in Fig. 1.

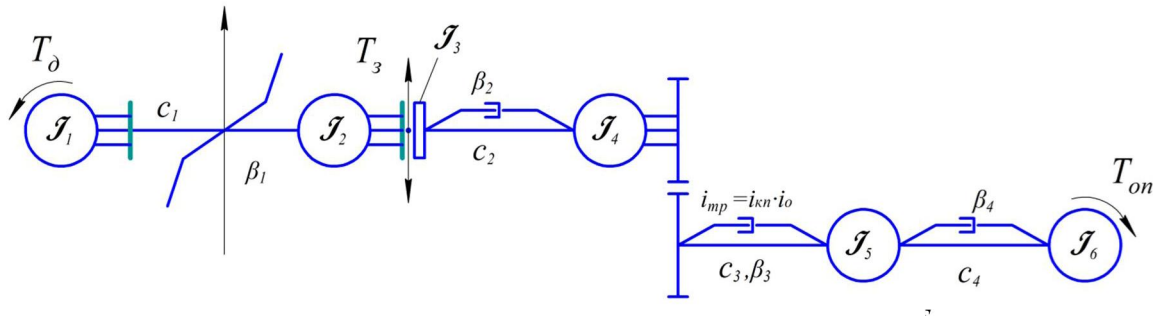


Fig. 1. Dynamic model of the front-wheel drive car with a DMF

Oscillatory processes in powertrain parts during acceleration will be described by a system of the following six differential equations [10], adding to them a refined description of the dependence of power and torque on the angular velocity of the crankshaft and uneven rotation:

$$\begin{cases} J_1 \ddot{\varphi}_1 = T_0 - c_1 (\varphi_1 - \varphi_2) - \beta_1 (\dot{\varphi}_1 - \dot{\varphi}_2); \\ J_2 \ddot{\varphi}_2 = c_1 (\varphi_1 - \varphi_2) + \beta_1 (\dot{\varphi}_1 - \dot{\varphi}_2) - T_3; \\ J_3 \ddot{\varphi}_3 = T_3 - c_2 (\varphi_3 - \varphi_4) - \beta_2 (\dot{\varphi}_3 - \dot{\varphi}_4); \\ J_4 \ddot{\varphi}_4 = c_2 (\varphi_3 - \varphi_4) + \beta_2 (\dot{\varphi}_3 - \dot{\varphi}_4) - c_3 \left(\varphi_4 \frac{1}{i_{mp}} - \varphi_5 \right) - \beta_3 \left(\frac{1}{i_{mp}} - \dot{\varphi}_5 \right); \\ J_5 \ddot{\varphi}_5 = c_3 \left(\varphi_4 \frac{1}{i_{mp}} - \varphi_5 \right) + \beta_3 \left(\dot{\varphi}_4 \frac{1}{i_{mp}} - \dot{\varphi}_5 \right) - c_4 (\varphi_5 - \varphi_6) - \beta_4 (\dot{\varphi}_5 - \dot{\varphi}_6); \\ J_6 \ddot{\varphi}_6 = c_4 (\varphi_5 - \varphi_6) + \beta_4 (\dot{\varphi}_5 - \dot{\varphi}_6) - T_{on}, \end{cases} \quad (1)$$

The figure shows: J_1 is a total moment of inertia of the internal combustion engine moving parts: crankshaft, pistons, connecting rods and a DMF primary mass; J_2 is the consolidated moment of inertia of a

DMF secondary mass and attached clutch components; J_3 is total moment of inertia of the clutch disc and the gearbox primary shaft mass; J_4 is consolidated moment of inertia of a gearbox secondary shaft and final drive components; J_5 is total moment of inertia of the hubs and rims of the wheels; J_6 is moment of inertia related the car's wheel mass; $c_1=2c_{sp}R^2$ is consolidated stiffness of the DMF sprung elements [10]; c_{sp} is linear stiffness of DMF springs mounted on a circle of R radius; c_2, c_3, c_4 is stiffness related, respectively, to gearbox components, axles, tires; $\beta_1, \beta_2, \beta_3, \beta_4$ are energy dissipation coefficients in DMF, gearbox, transmission, tires, respectively; $\varphi_1, \varphi_2, \varphi_3, \varphi_4, \varphi_5, \varphi_6$ are the angles of rotation of the masses $J_1, J_2, J_3, J_4, J_5, J_6$, respectively; T_d is torque of the internal combustion engine; T_{3q} is the clutch moment of friction; T_{on} is the moment of the movement resistance related to the car wheels.

Effective torque of internal combustion engine during acceleration will be presented in the form:

$$T_e = aN_1\omega_e + bN_2\omega_e^2 + cN_3\omega_e^3, \quad (2)$$

where a, b, c and N_1, N_2, N_3 coefficients are determined by the formulas:

$$a = \frac{k_\omega k_T (2 - k_\omega) - 1}{k_\omega (2 - k_\omega) - 1}, \quad (3)$$

$$b = -\frac{2k_\omega (k_T - 1)}{k_\omega (2 - k_\omega) - 1}, \quad (4)$$

$$c = \frac{k_\omega^2 (k_T - 1)}{k_\omega (2 - k_\omega) - 1}, \quad (5)$$

$$N_i = N_{\max} / \omega_N^i. \quad (6)$$

where k_T, k_ω are engine adaptation coefficients related to torque and angular velocity (speed); N_{\max} is maximum power, ω_N is angular velocity at maximum power.

Internal combustion engine crankshaft torsional oscillations will be reflected by multiplying torque curve by sine wave function with appropriate frequency.

Car movement resistance moment related to the wheels:

$$T_{on} = G_a r \left[\left(f_0 + k_f (\dot{\varphi}_6 r)^2 \right) \cos \gamma + \sin \gamma \right] + \delta m_a r^2 \ddot{\varphi}_6, \quad (7)$$

where G_a is vehicle weight, r is wheel dynamic radius, f_0 is low speed rolling resistance coefficient, k_f is the coefficient taking into account the increase in rolling resistance with increasing vehicle speed, γ is the inclination angle, δ is the coefficient taking into account the impact inertia of rotating masses of the car powertrain at its accelerated movement, m_a is the car mass, and $\dot{\varphi}$ and $\ddot{\varphi}$ are accordingly angular speed and angular acceleration of car wheels.

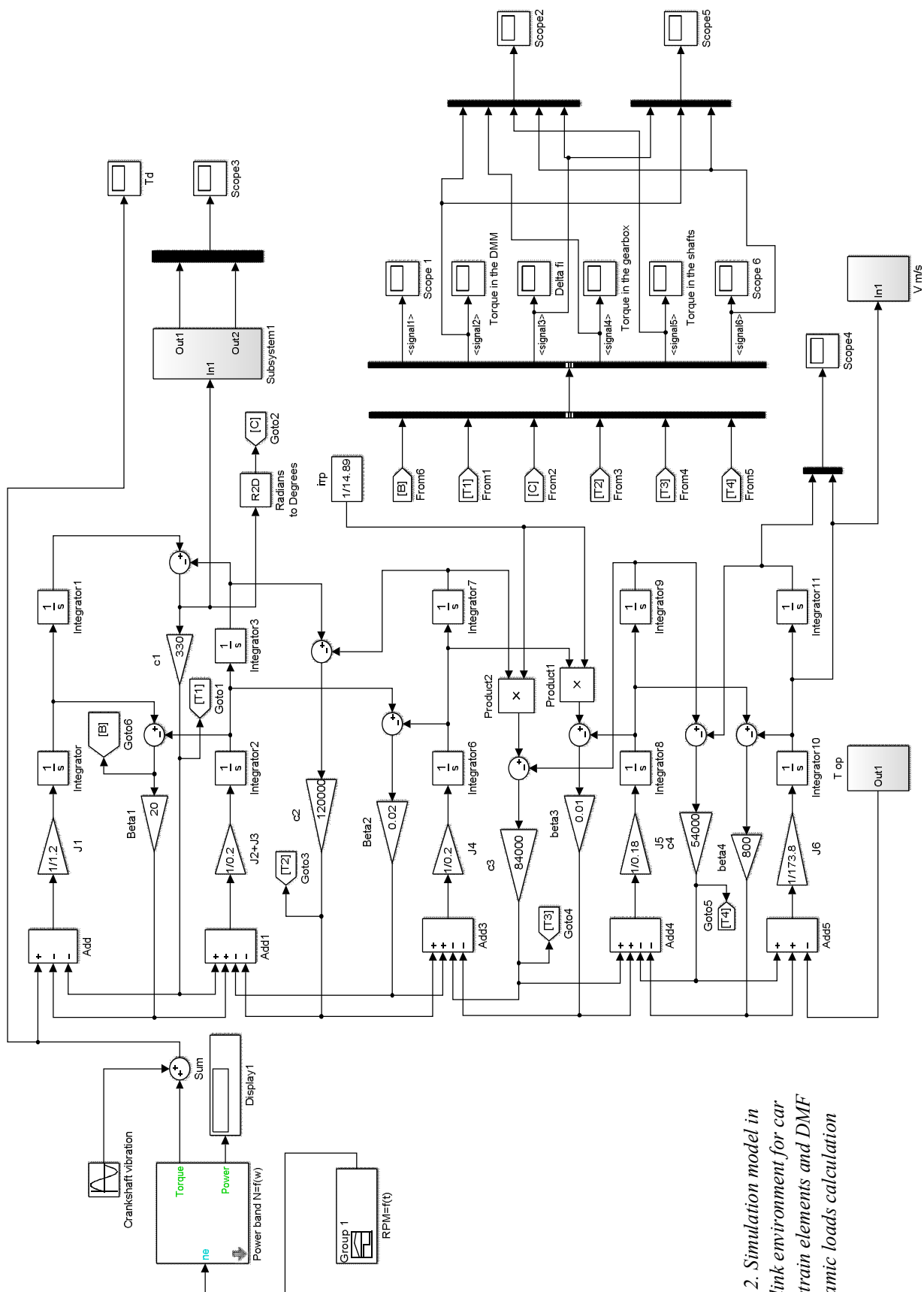
The tension in DMF springs coils in the process of dynamic loading during the car movement will be determined by a known formula:

$$\tau_{\max} = \frac{16(\varphi_1 - \varphi_2) c_{inp} R_{icep}}{\pi d_i^3} \left(\frac{4m-1}{4m-4} + \frac{0,615}{m} \right), \quad (8)$$

where c_{inp} is the linear stiffness of the certain spring, R_{icep} is the average radius of the certain spring, d_i is the wire diameter of the certain spring, $m = 2R_{icep}/d_i$ is the correction factor for the certain spring.

The above dependences are a mathematical model of the car powertrain comprises a DMF in the process of movement start, the implementation of which allows to investigate the change in dynamic loads in the spring elements of a DMF and powertrain shafts at different intensities.

Simulation model. An appropriate simulation model was developed to calculate the dynamic loads on the elastic links of the DMF and the drive shaft when starting the car in first speed in the Simulink environment (Fig. 2). The subsystem for calculating the dynamic loads on the drive elements of the car and DMF is shown in Fig. 3, and a, b, c coefficients calculation in Fig. 4.



Using the developed model, the process of a car beginning the movement on a dry asphalt road was simulated with a sharp engagement of a clutch in first gear and different duration of power gain and increasing torque of an internal combustion engine. This mode of operation is typical for robotic gearboxes with two dry clutches without a torque converter and is formed by the degree of pressure on the accelerator pedal by the driver.

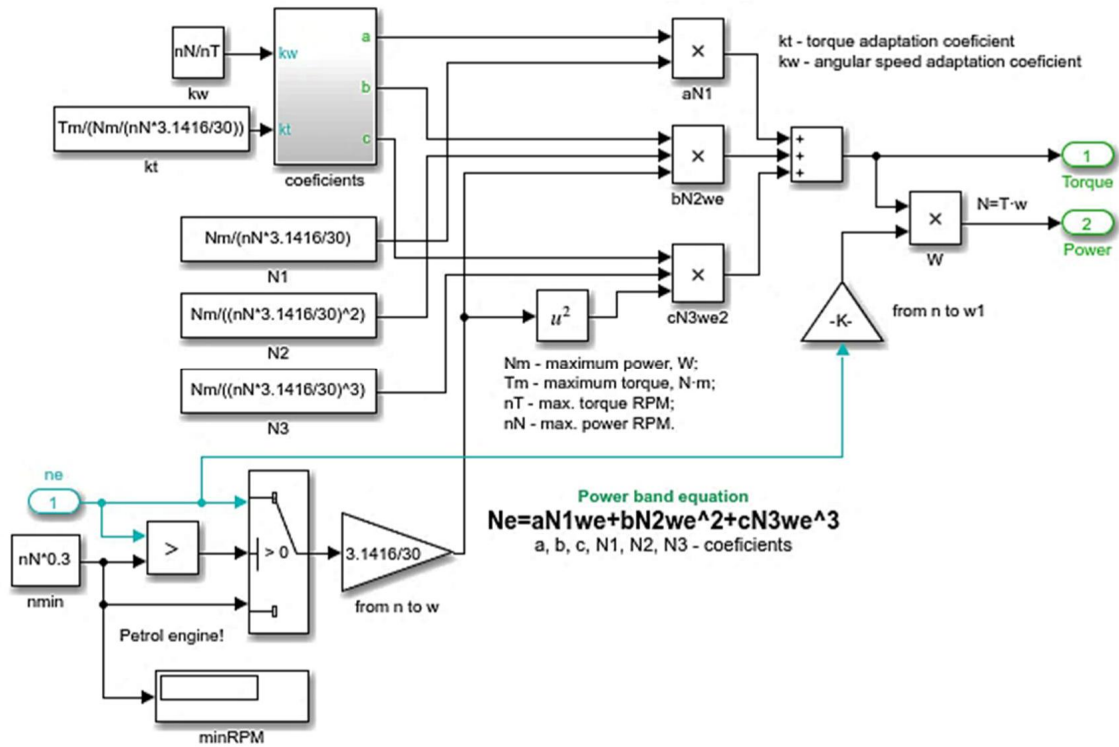


Fig. 3. Simulation model for the ICE torque and power calculating

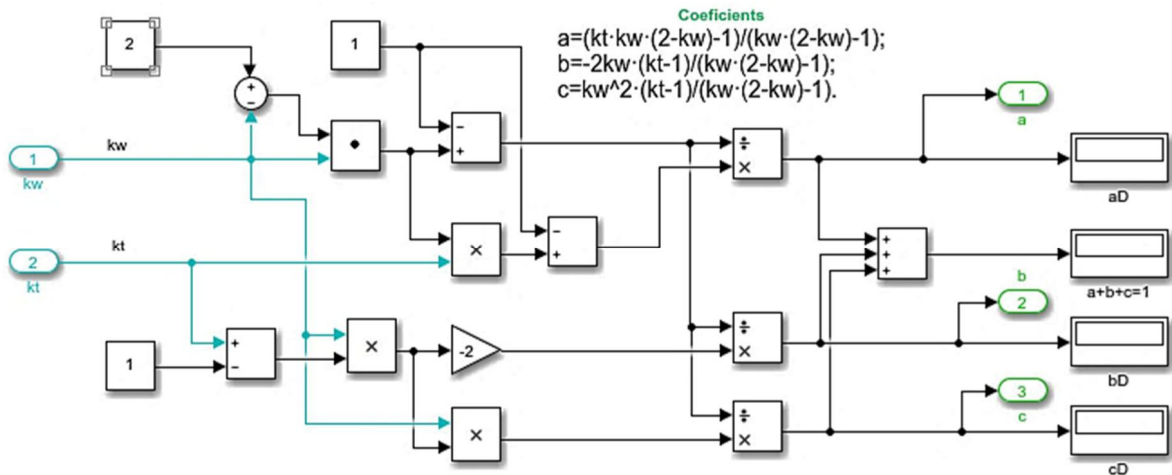


Fig. 4. Simulation model for the a , b , c coefficients calculating

Simulation results and their analysis. The research was carried out with parameters of powertrain dynamic model (see Fig. 1), corresponding to the M1 vehicle category, in particular: the maximum engine power is $N_{max} = 75$ kW which achieves at 4400 rpm, maximum torque $T_0 = 250$ N·m which achieves at 1500 rpm, the variable component of the torque of the internal combustion engine was set at 10 N·m. Coefficients of the engine adaptability by torque and angular velocity (speed) were selected as follows: $k_T = 1.54$; $k_w = 2.93$. Combined moment of inertia of internal combustion engine moving parts

$J_1 = 1.2 \text{ kg}\cdot\text{m}^2$; combined moment of inertia of the DMF secondary mass and attached clutch components $J_2 = 0.1 \text{ kg}\cdot\text{m}^2$; the total moment of inertia of the movable gearbox components $J_3 = 0.1 \text{ kg}\cdot\text{m}^2$; the total moment of inertia of the gearbox secondary shaft and final drive components $J_4 = 0.2 \text{ kg}\cdot\text{m}^2$; the combined moment of inertia of the wheels hubs and rims $J_5 = 0.18 \text{ kg}\cdot\text{m}^2$; reduced moment of inertia of the car mass to the wheel $J_6 = 173.8 \text{ kg}\cdot\text{m}^2$. Total torsional rigidity: gearbox components $c_2 = 120.000 \text{ N}\cdot\text{m}/\text{rad}$; drive axles $c_3 = 84000 \text{ N}\cdot\text{m}/\text{rad}$; rims with tires $c_4 = 27000 \text{ N}\cdot\text{m}/\text{rad}$. Energy dissipation coefficients: in DMF $\beta_1 = 4 \text{ N}\cdot\text{s}/\text{m}$; in gearbox components $\beta_2 = 0.02 \text{ N}\cdot\text{s}/\text{m}$; in drive axles $\beta_3 = 0.01 \text{ N}\cdot\text{s}/\text{m}$; in tires $\beta_4 = 800 \text{ N}\cdot\text{s}/\text{m}$. The road inclination angle $\gamma = 0^\circ$. Total transmission ratio = 14.89.

Three variants of car movement start are considered: the first – slow acceleration for 3 s with a change in the number of internal combustion engine crankshaft revolutions from 1400 to 1500 rpm; the second: when reaching the maximum number of revolutions of the internal combustion engine $n_{08} = 4400 \text{ rpm}$ for 2 s; the third – when reaching the maximum number of revolutions of the internal combustion engine for 4 s. The simulation results are shown in Fig. 5 – Fig. 13.

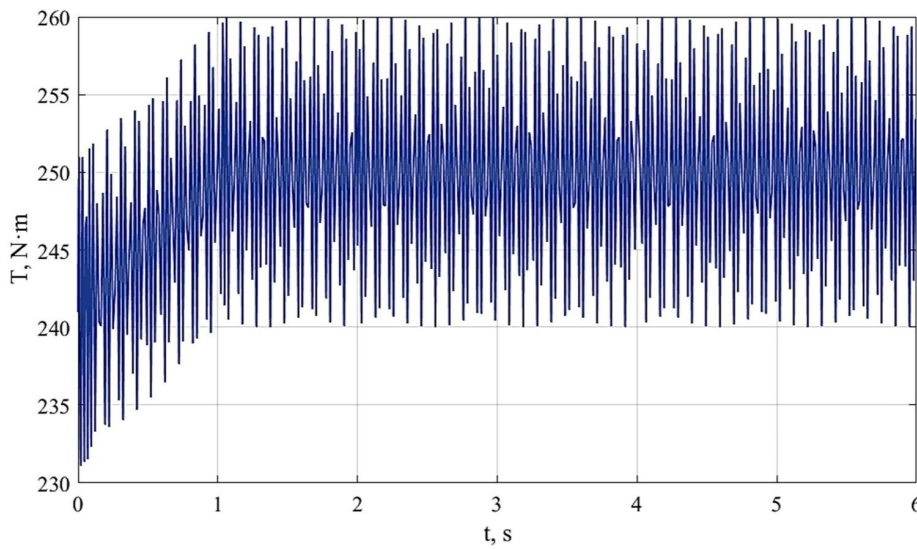


Fig. 5. Change of torque of the internal combustion engine from time:
 $n_1 = 1400 \text{ rpm}$, $n_2 = 1500 \text{ rpm}$, $t_p = 2 \text{ s}$

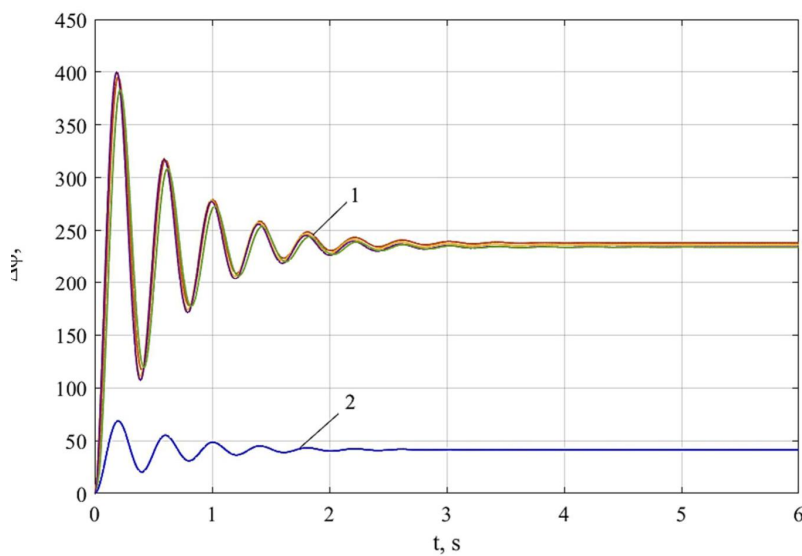


Fig. 6. Change of torque in DMF components and transmission (1)
 and the relative twist angle of DMF masses (2) from time to time:
 $n_1 = 1400 \text{ rpm}$, $n_2 = 1500 \text{ rpm}$, $t_p = 2 \text{ s}$

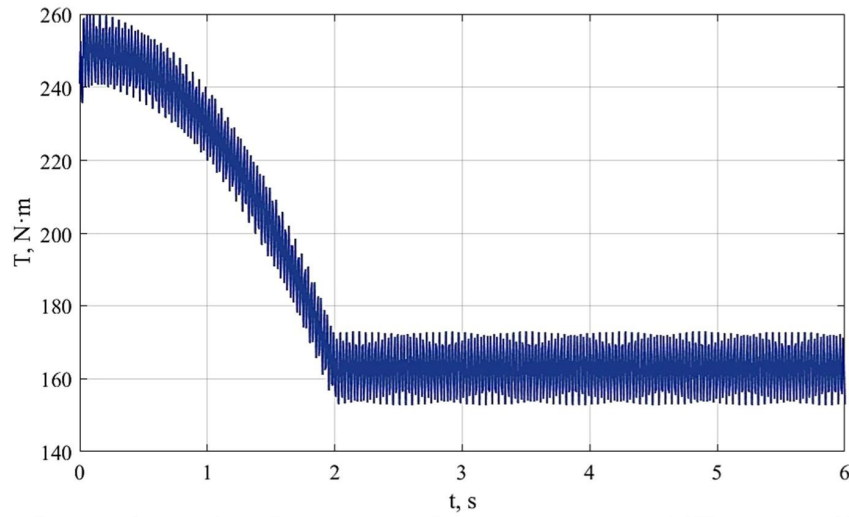


Fig. 7. Torque changes of internal combustion engine from time to time:
 $n_1 = 1400 \text{ rpm}$, $n_2 = 4400 \text{ rpm}$, $t_p = 2 \text{ s}$

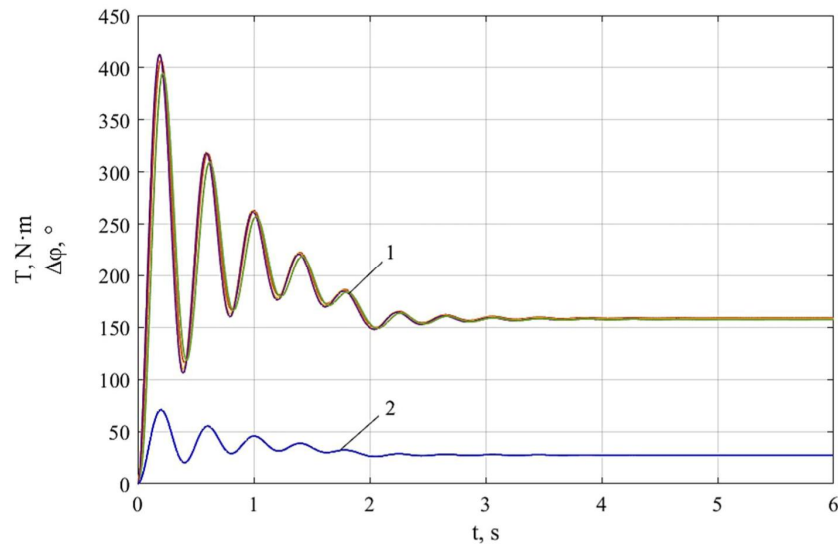


Fig. 8. Torque changes in DMF components and transmission (1)
 and the relative twist angle of DMF masses (2) from time to time: $n_1 = 1400 \text{ rpm}$, $n_2 = 4400 \text{ rpm}$, $t_p = 2 \text{ s}$

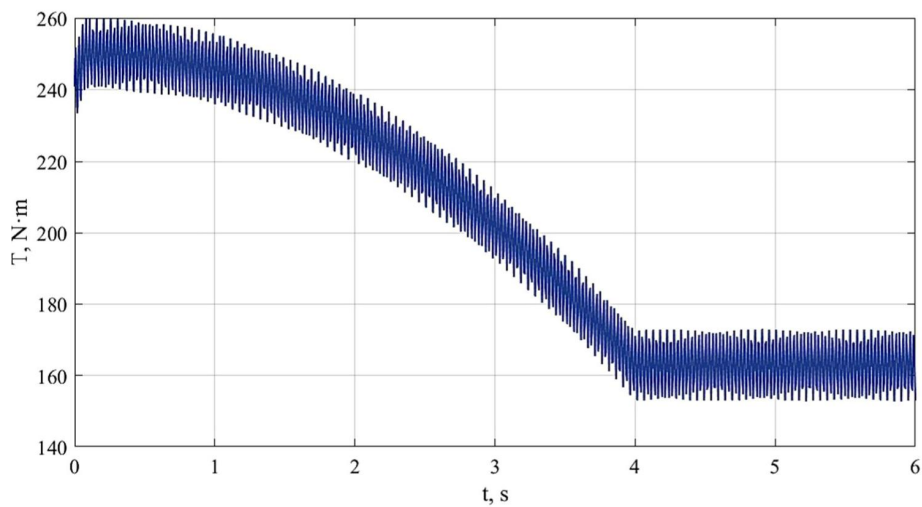
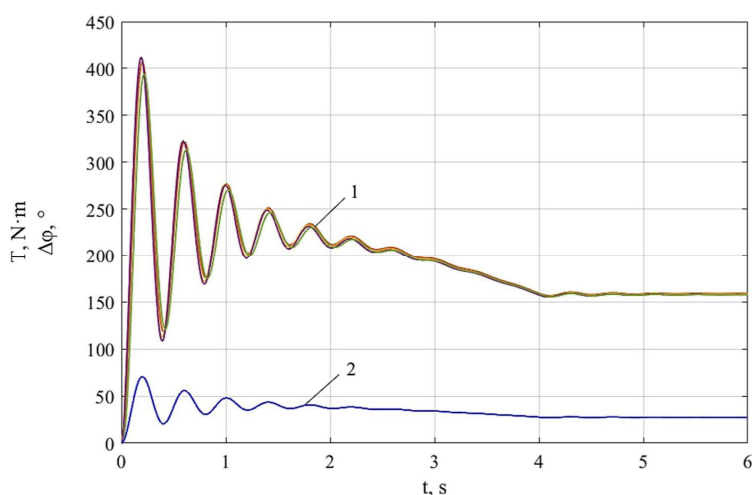


Fig. 9. Torque changes of the internal combustion engine from time:
 $n_1 = 1400 \text{ rpm}$, $n_2 = 4400 \text{ rpm}$, $t_p = 4 \text{ s}$

Fig. 10. Torque changes related to the DMF components and transmission (1) and the relative twist angle of the DMF masses (2) from time to time: $n_1 = 1400$ rpm, $n_2 = 4400$ rpm, $t_p = 4$ s



Analyzing the course of oscillating processes in the drive of the car in different movement modes of the car movement start in Fig. 6–10, we see that with a slow increase in the number of revolutions of the crankshaft from 1400 to 1500 rpm, until the torque of the internal combustion engine reaches its maximum value of 250 N·m (see Fig. 6), the amplitude of the elastic moment in the powertrain components increases sharply to 400 N·m (see Fig. 7, curves 1), and the DMF masses angle of relative torsion is close to the maximum allowable value. After 2 s, the oscillations of the elastic moment in the transmission components are damped, stabilizing at 240 N·m.

During intensive car starting, when the speed of crankshaft rotation for 2 s increases from 1400 to 4400 rpm, the torque of the internal combustion engine first increases to 250 N·m and decreases rapidly, stabilizing for 2 s to 165 N·m (see Fig. 7). At the same time, the elastic moment in the powertrain components initially increases to 400 N·m, but quickly attenuates and stabilizes for 2 s at 165 N·m, which has a positive effect on the load on the powertrain components.

Increasing the duration of the engine crankshaft speed to 4 s (see Fig. 9) only leads to an increase in the load time of the powertrain components with cyclic and steady torque (see Fig. 10), which worsens the powertrain operating conditions.

DMF springs perceive a similar load type. In Fig. 11–13 shows the dependences of tension changes in the coils of DMF springs: 1 depicts springs of smaller diameter, 2 depicts springs of larger wire diameter.

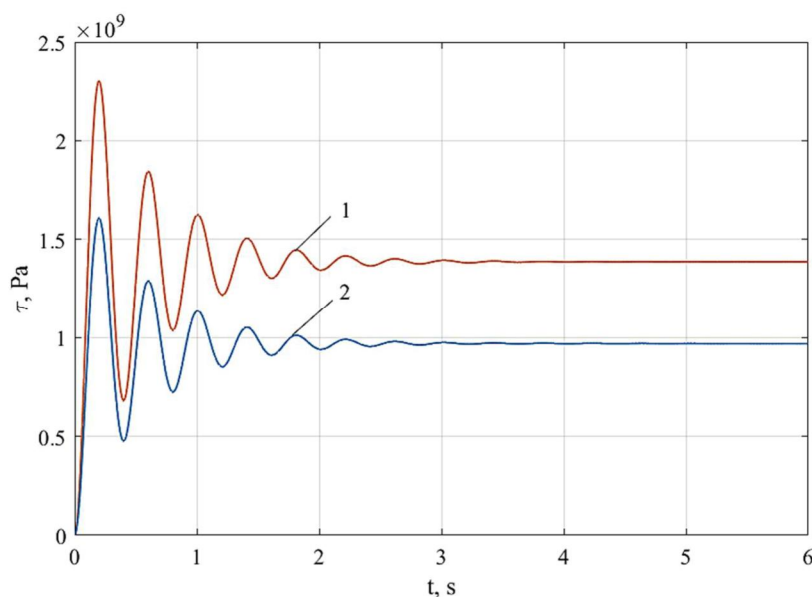


Fig. 11. Change of tensions in DMF components from time: $n_1 = 1400$ rpm, $n_2 = 1500$ rpm, $t_p = 2$ s

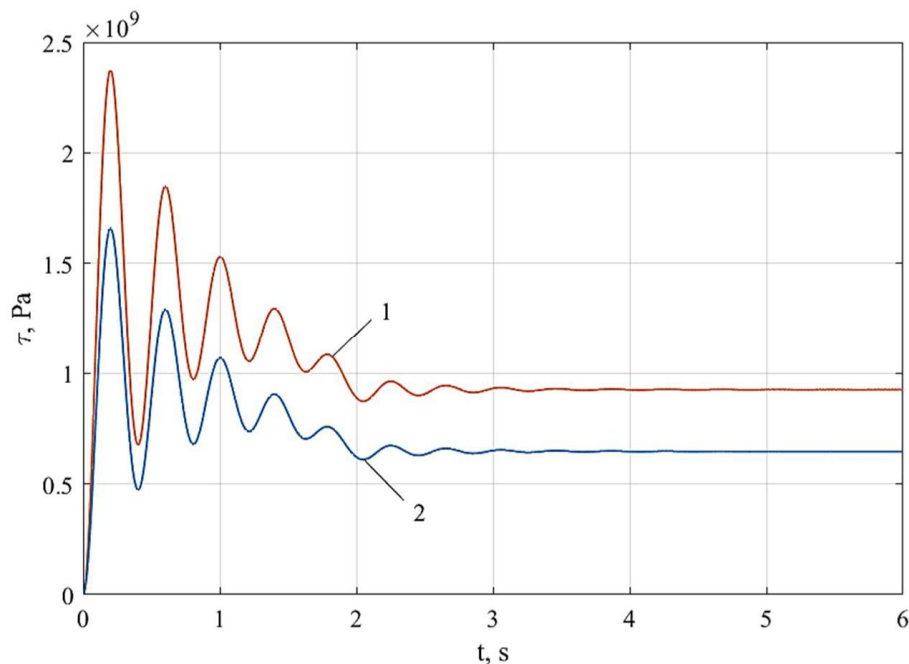


Fig. 12. Change of tensions in DMF components from time:
 $n1 = 1400 \text{ rpm}$, $n2 = 4400 \text{ rpm}$, $tp = 2 \text{ s}$

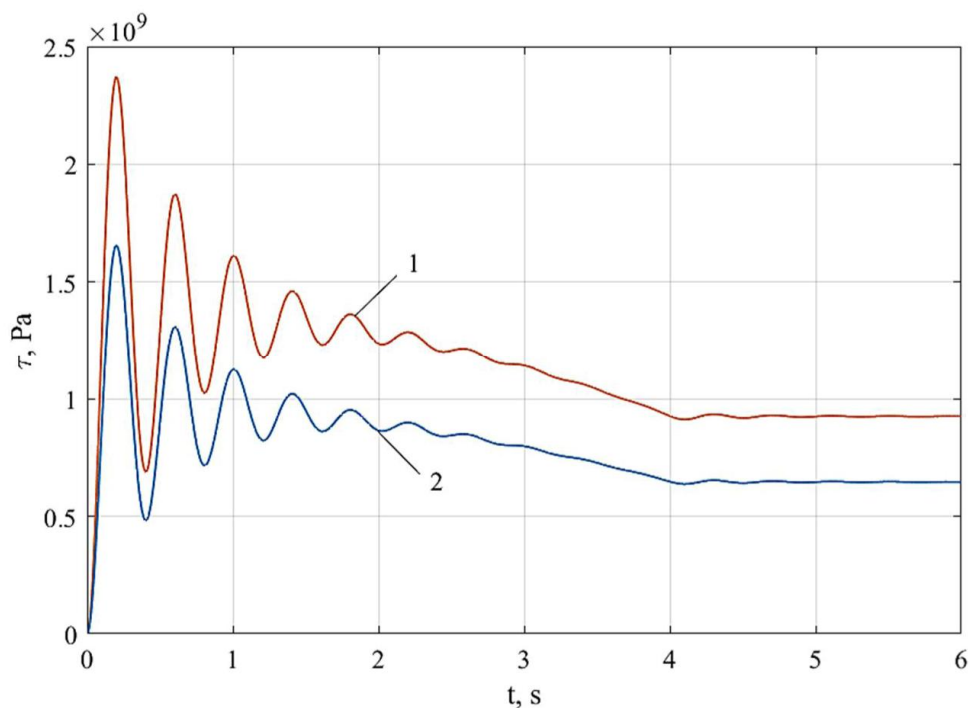


Fig. 13. Change of tension in DMF components from time:
 $n1 = 1400 \text{ rpm}$, $n2 = 4400 \text{ rpm}$, $tp = 4 \text{ s}$

Analysis of graphs in Fig. 11–13 indicates that increasing the car acceleration speed when starting leads to a decrease in the duration of DMF loading springs cyclic forces, and therefore increases their life, reducing the risk of fatigue failure.

Therefore, to increase the DMF service life, it is advisable to accelerate a car when moving intensively, bringing the internal combustion engine number of crankshaft revolutions to a value that is located at the extremum of engine torque on its external speed characteristic, followed by switching to the next gear.

4. CONCLUSIONS

1. Developed a mathematical and MATLAB Simulink simulation model of a car powertrain comprising a DMF in the period of starting, which takes into account the dependence of torque and power of the internal combustion engine on the number of crankshaft revolutions and uneven rotation, car powertrain inertial and rigid parameters, road resistance.

2. On the example of a M1 vehicle category studied the effect of the acceleration intensity when starting in first gear for different durations to achieve a certain number of crankshaft revolutions and, accordingly, the torque generated by an internal combustion engine.

3. It is established that when starting the car in first gear, the maximum sprung components load of the DMF and transmission occurs at the initial moment of engagement of the clutch and exceeds the maximum effective torque of the internal combustion engine 1.6 times, has a pronounced oscillatory nature and, stabilizes. During smooth acceleration of the car, when the torque of the internal combustion engine doesn't reach its maximum value of 250 N·m, the elastic moment in the powertrain components is stabilized at 230 N·m. During intensive acceleration and transition through the extremum on the curve of the torque of the internal combustion engine on the number of revolutions of the crankshaft, the maximum load on the DMF sprung components and transmission initially doesn't change significantly, but reduces the duration of oscillations and the elastic moment of 165 N·m after attenuation of oscillations.

4. To increase the DMF service life, it is advisable to accelerate the car when moving intensively, bringing the number of revolutions of the internal combustion engine to a value that is located at the extremum of the engine torque on its performance characteristics, followed by switching to the next gear.

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Received 05.04.2022; Accepted in revised form 18.04.2022.

ВПЛИВ ІНТЕНСИВНОСТІ РОЗГОНУ АВТОМОБІЛЯ НА НАВАНТАЖЕНІСТЬ ЕЛЕМЕНТІВ ДВОМАСОВОГО МАХОВИКА І ТРАНСМІСІЇ

Анотація. Сучасні високомоментні низькообертові двигуни внутрішнього згорання (ДВЗ) генерують крутильні коливання близькі за частотою збурення до власних частот коливань ланок коробок передач (КП). Ефективне поглинання таких коливань вимагає нового конструктивного виконання демпфера крутильних коливань між ДВЗ і КП, що реалізовано у вигляді двомасового маховика (ДММ). Однією з головних причин виходу з ладу ДММ є руйнування його пружних ланок. У статті розроблено математичну й симуляційну (у середовищі *Matlab Simulink*) моделі приводу автомобіля з ДММ у період рушання з місця, яка враховує залежність величини крутного моменту і потужності ДВЗ від кількості обертів колінчастого вала та нерівномірність його обертання, інерційні та жорсткісні параметри приводу автомобіля, опір дороги. Встановлено, що при русі автомобіля з місця на першій передачі максимальне навантаження на пружні ланки ДММ і трансмісії виникає у початковий момент вмикання зчеплення і перевищує максимальний ефективний крутний момент ДВЗ в 1,6 рази, має виражений коливальний характер і в міру розгону автомобіля стабілізується. При плавному розгоні автомобіля, коли крутний момент ДВЗ досягає, але не перевищує, свого максимального значення 250 Н·м, пружний момент у ланках приводу стабілізується на рівні 230 Н·м. За інтенсивного розгону та переходу через екстремум на кривій залежності крутного моменту ДВЗ від кількості обертів колінчастого вала максимальне навантаження на пружні ланки ДММ і трансмісії у початковий момент за величиною істотно не змінюється, але зменшуються тривалість протікання коливальних процесів і величина пружного моменту в ланках приводу до 165 Н·м після затухання коливань. Аналогічний характер зміни напружень спостерігається і в пружних ланках ДММ, що з часом призводить до їх втомного руйнування і виходу з ладу ДММ.

Для підвищення ресурсу ДММ доцільно розгін автомобіля при русі з місця здійснювати інтенсивно, доводячи кількість обертів до величини, яка розташована за екстремумом крутного моменту ДВЗ на його зовнішній швидкісній характеристиці, з подальшим перемиканням на наступну передачу.

Ключові слова: трансмісія, двомасовий маховик, динамічна модель, математична модель, симуляційна модель, крутний момент.